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# A review of heat pipe systems for heat recovery and renewable energy applications

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ABSTRACT

Advancements into the computational studies have increased the development of heat pipe arrangements, displaying multiphase flow regimes and highlighting the broad scope of the respective technology for utilization in passive and active applications. The purpose of this review is to evaluate current heat pipe systems for heat recovery and renewable applications utility. Basic features and limitations are outlined and theoretical comparisons are drawn with respect to the operating temperature profiles for the reviewed industrial systems. Working fluids are compared on the basis of the figure of merit for the range of temperatures. The review established that standard tubular heat pipe systems present the largest operating temperature range in comparison to other systems and therefore offer viable potential for optimization and integration into renewable energy systems.



**Contents**



## 1. Introduction

A heat pipe is a simple device of very high thermal conductivity with no moving parts that can transport large quantities of heat efficiently over large distances fundamentally at an invariable temperature without requiring any external electricity input. A heat pipe is essentially a conserved slender tube containing a wick structure lined on the inner surface and a small amount of fluid such as water at the saturated state. It is composed of three sections: the evaporator section at one end, where heat is absorbed and the fluid is vaporized; a condenser section at the other end, where the vapor is condensed and heat is rejected; and the adiabatic section in between, where the vapor and the liquid phases of the fluid flow in opposite directions through the core and the wick,

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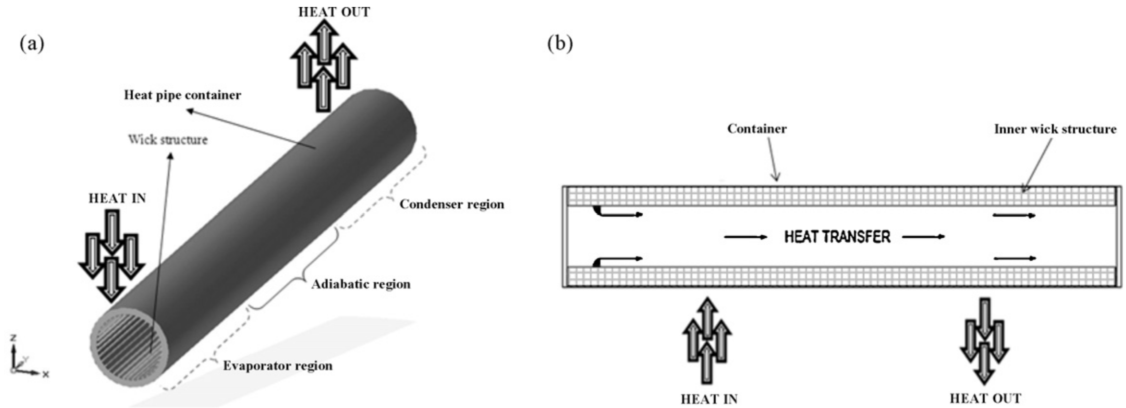



**Fig. 1.** Basic working principle associated with a heat pipe (a) isometric view and (b) sectional view.

respectively, to complete the cycle with no significant heat transfer between the fluid and the surrounding medium.

The operating pressure and the type of fluid inside the heat pipe depend largely on the operating temperature of the heat pipe. For example, if a heat pipe with water as a working fluid is designed to remove heat at 343 K, the pressure inside the heat pipe must be maintained at 31.2 kPa, which is the boiling pressure of water at this temperature. Though water is a suitable fluid to utilize in the moderate temperature range encountered in electronic equipment, various other fluids are used in the manufacturing of heat pipes to allow them to be used in cryogenic as well as high-temperature applications. Another characteristic while selecting the working fluid is the property of surface tension, which must be high in order to increase the capillary effect and being compatible with the wick substance, as well as being chemically stable, readily available, non-toxic and inexpensive [1]. Fig. 1 displays the basic working sections of a heat pipe.

Heat pipes are utilized in a wide variety of applications which encounter temperature variations in a heat transfer process. The effectual thermal conductivity of a heat pipe facilitates heat to be transported at high efficiency over large distances. Consequently, heat pipes have been expansively used in various energy storage systems due to their suitability in the role of heat delivery and passive operation. The unique method of operation of heat pipes including phase change materials (PCMs) provide a better efficiency pattern over conventional heat exchangers in major operations including temperature stratification in hot water storage tanks. Another general utility of heat pipes include solar collectors where it allows static or flowing water to be heated by the method of transferring the solar thermal energy directly from the sun [2].

## 2. The role of heat pipes in heat recovery and energy conservation

The demand for utilizing heat pipes in renewable energy systems along with building heat recovery, highlighting novel concepts and requirements is increasing. Several terrestrial applications ranging from solar concentrators to heat exchangers make use of heat pipes for higher and more efficient heat transfer rates. Heat pipes offer distinct advantages over other thermal transfer apparatus due to its passive and compact method of operation along with the various commercial sizes available ranging from micro to a more extensive array making the device suitable for most applications requiring a temperature differential.

El-Baky and Mohamed [3] investigated the overall effectiveness of utilizing heat pipe heat exchangers for heat recovery through external air-conditioning systems in buildings in order to reduce the cooling load. The thermal performance of the system was analyzed for varying fresh air inlet mass flow rates and temperatures stream. A mathematical model was developed based on the experimental set-up which included the two air ducts of 0.3 m × 0.22 m sectional areas along with the heat pipe arrangement comprising of 25 copper tubes with the evaporator and condenser section of 0.2 m and the adiabatic section of 0.1 m respectively. R-11 was used as a working fluid at a saturation temperature of 303 K. The findings of the study indicated that effectiveness and heat transfer rates are increased with the increase in fresh air inlet temperature. The study also revealed that the mass flow rate ratio has a significant effect of temperature change of fresh air and heat recovery rate is increased by approximately 85% with the increase in fresh air inlet temperature. Fig. 2 describes the schematic of the heat exchanger.

Noie-Baghban and Majideian [4] carried out work on the design and build of a heat pipe arrangement to be installed in a heat pipe heat exchanger for the purpose of heat recovery in hospital and laboratory buildings where high air change is a primary requirement. The experimental apparatus include a test-rig comprising of two fans to deliver a flow rate of 0.103 $m^3$/s through evaporator and condenser. Eight copper pipes with an outside diameter of 15 mm, inside diameter of 9 mm and length of 600 mm were utilized along with three types of wicks including the 50 mesh nickel, 250 mesh nickel and 100 mesh stainless steel. The figure of merit of the type of working fluid was established. K-type thermocouples were used for temperature measurements. A mathematical model was established to validate the experimental findings. The work concluded a good correlation between the mathematical and experimental results with respect to the heat transfer rate in the evaporator section of 100 W. Further, the study highlighted the importance of utilizing finned heat pipes and increasing the number of rows along with insulation capability in having a major impact in increasing the overall effectiveness of the system.

Various renewable applications are highlighted in this review in order to understand the role of heat pipes to a broader extent. A gas–gas heat pipe heat exchanger consists of a collection of similar heat pipes aligned in a tubular arrangement either vertically, horizontally or aligned at an angle. The evaporation and condensation working principal of the device influences the heat transfer from the countercurrent gas stream which recovers the heat and transports it to the pre-heated air stream. Heat pipe heat exchangers are very useful in industrial heat recovery applications due to its static operation and limited auxiliary power requirements along with its entirely reversible process.

Yau and Ahmadzadehtalatapeh [5] reviewed the utility of horizontal pipe heat exchangers as an energy recovery unit in air conditioning systems in tropical climates. The review included literature from previously published work on the vertical and

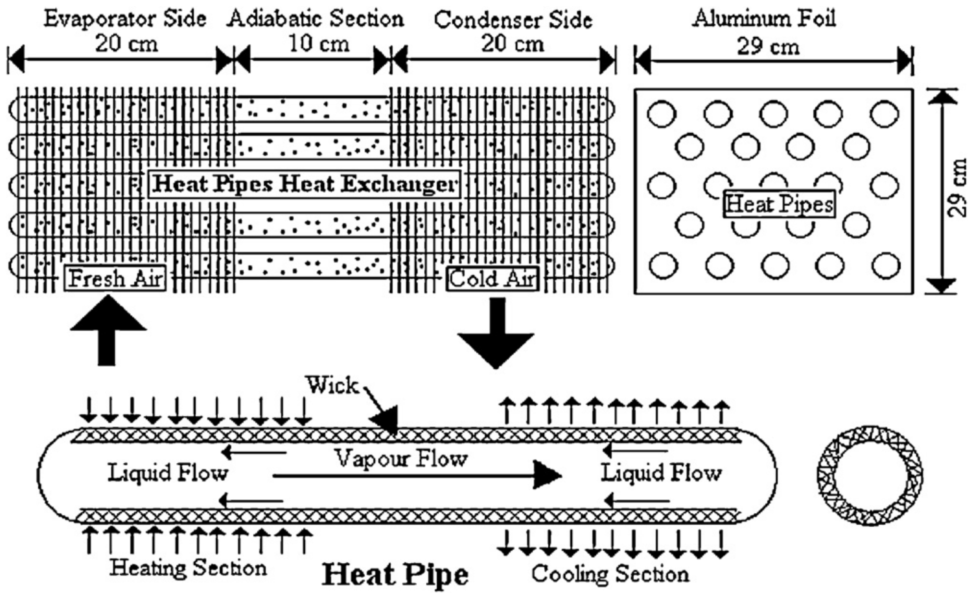



**Fig. 2.** Heat pipe heat exchanger design [3].

horizontal orientations of heat pipes respectively. The work concluded that the application of horizontal heat pipe heat exchangers for both orientations in terms of dehumidification purposes and energy saving is recommended for tropical climates as a highly efficient heat recovery unit. The work further highlighted the transient simulation of installing double heat pipe heat exchanger units in heating, ventilation and air-conditioning systems for reducing energy consumption rates in tropical climatic behavior as displayed in Fig. 3.

One of the most widespread commercial uses of heat pipes is associated with solar collectors in order to transfer the direct and diffuse solar radiation to the water stream. Hussein et al. [6] carried out test work on the comparison of three cross-sectional geometries of wickless heat pipes with varying fill ratios in order to understand the impact of its performance on flat plate solar collectors in Cairo, Egypt. The manufacturing group comprised of heat pipe cross-sections which included circular, elliptical and semi-circular arrangement. Experiments were conducted on the group by incorporating the heat pipes into the solar collector array and the comparison results indicated that the elliptical design gave a better performance at 10% water fill ratios with the circular cross-section design proving optimum at 20% water fill ratio respectively.

Rittidech and Wannapakne [7] carried out extensive work on determining the overall performance capability of a system comprising of a Closed-End Oscillating Heat Pipe (CEOHP) incorporated into a flat plate solar collector. The thermocouple based experimental test apparatus was inclined at 18 degrees and comprised of a 2 m zinc sheet coupled with 70 m of CEOHP copper tubes. The working fluid comprised of R134a at an initial fill ratio of 50%. Fig. 4 describes the schematic of the test-rig where A1–A6 is the thermo-junction on the collecting plate and G1–G2 is the thermo-junction position on the glass plate respectively. A numerical model was built to calculate the performance of the system with respect to the plate temperature and ambient temperature and an overall thermal efficiency of 62% was obtained. The study highlighted the advantages of using CEOHP system in comparison to conventional heat pipe systems on solar collectors in terms of minimal corrosion rate and elimination of freezing during winter.

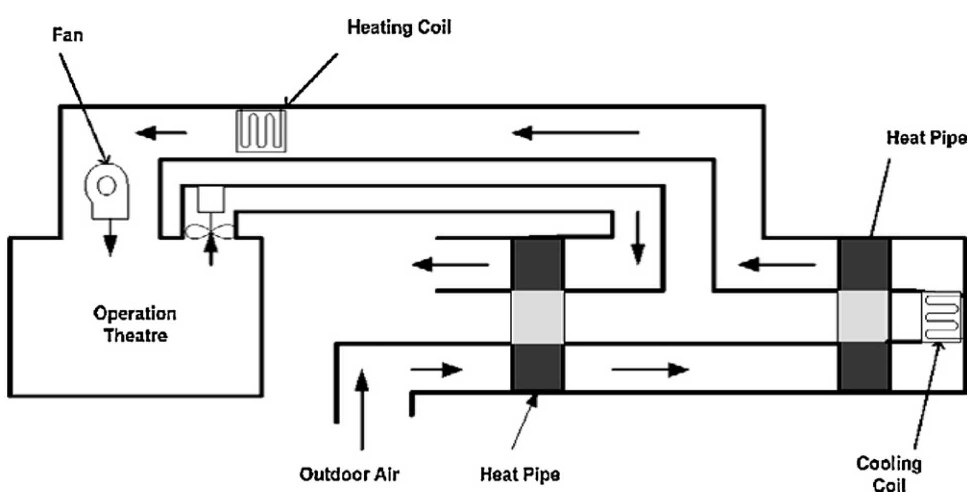



**Fig. 3.** Schematic of double heat pipe systems in the heating, ventilation and air-conditioning [5].

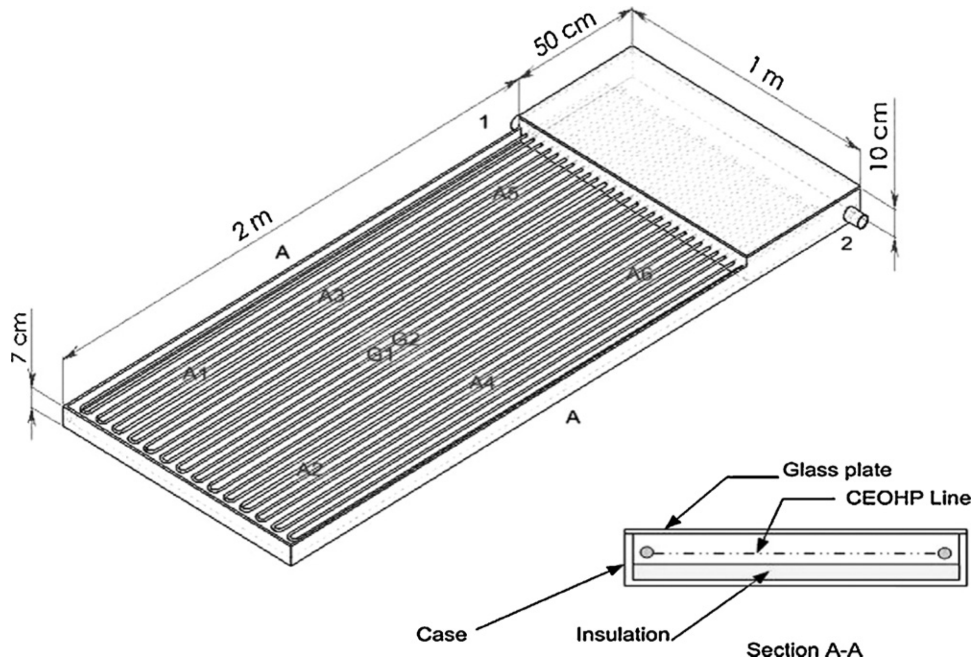



**Fig. 4.** Outline of the thermo-junction positions on collecting and glass plates in the test rig [7].

## 3. Conventional heat pipe systems

Various types of heat pipes are commercially available, in terms of the method of liquid transport from the condenser to the evaporator and functionality. This review provides a source of information based on the current published literature on the different types of existing heat pipes which are utilized for a variety of applications requiring moderate to high temperature fluctuations.

### 3.1. Tubular heat pipes

Conventional tubular heat pipes as displayed in Fig. 5 are the most uncomplicated and accepted type of passive heat transfer devices commercially for use in many terrestrial applications for heat transport over variable distances. The standard operational principle is based on capillary action and the performance is measured in equivalent thermal conductivity. These types can also be used as heat spreaders to isothermalize apparatus where homogeneous temperature patterns are preferred.

Liao et al. [8] analyzed the thermal performance of a smooth carbon steel-water heat pipe in comparison to its internally finned equivalent. Various influencing parameters including the inclination angle, working temperatures and heat flux formed the basis of the investigation. The experimental set-up comprised of a fiber glass coated carbon steel pipe with a flat band heater for providing heat flux to the evaporator section. The apparatus was placed on an adjustable workbench for alteration of inclination angles and thermocouples were linked to the data logging system for output results. The work revealed that under experimental conditions, the heat transfer coefficient of the internally finned heat pipe was increased by 50–100% in comparison to the smooth heat pipe respectively.

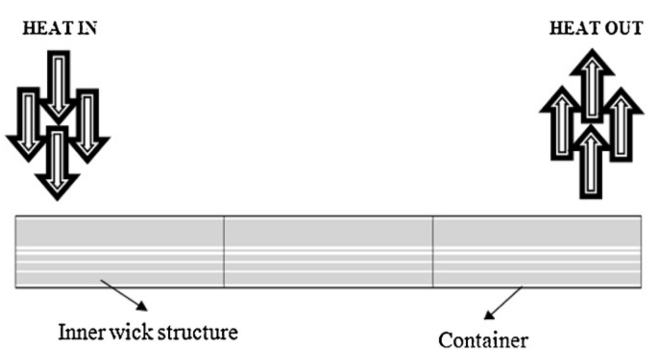



**Fig. 5.** Schematic of a tubular heat pipe.

Joudi and Witwit [9] carried out work to improve the thermal performance of gravity assisted conventional wickless heat pipes. Experimental study was carried out on the modified copper heat pipe with the introduction of an adiabatic separator. The heat pipe under test was fixed in a rig and coupled with several measuring devices including a digital ammeter and voltmeter in order to calculate the input power. The heat pipe was insulated with glass wool to minimize heat losses to the environment. The condenser flow rate was kept constant and the temperature was monitored at $23 \pm 2\,^{\circ}$C and the power input was increased steadily to obtain gradual thermocouple readings. The outcome from the study highlighted useful results with respect to the addition of an adiabatic separator in the heat pipe. The study revealed an approximate increase of 35% in heat transfer coefficient in comparison to conventional heat pipes. The investigation concluded that the addition of an adiabatic separator eradicated the effect of inclination angles above 45° and decreased the heat pipe working temperature respectively.

### 3.2. Variable conductance heat pipes

Variable Conductance Heat Pipes (VCHPs) are widely utilized in many applications including conventional electronics temperature control. A variable conductance heat pipe or gas-loaded heat pipe has the capability to maintain a device mounted at the evaporator at a near constant temperature, independent of the amount of power being generated by the device. The most familiar VCHP systems include passive or active feedback-controlled system, both having the capability to control the source of heat at the evaporator end. However, a greater temperature control is obtained using the active system than the comparable passive system. Fig. 6 displays

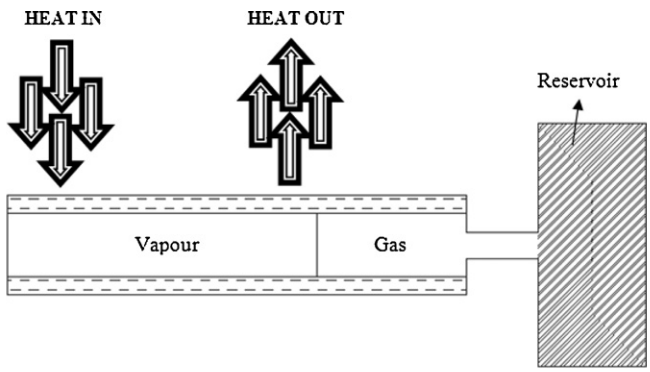



**Fig. 6.** Schematic of a cold-reservoir variable conductance heat pipe.

the schematic of a cold-reservoir variable conductance heat pipe [2].

Sauciuc et al. [10] analyzed the operation of a VCHP for controlling the temperature of a closed system arrangement of solar collectors. The experimental apparatus included a copper/water heat pipe outfitted with a cold reservoir and used air as the Non-Condensable Gas (NCG). The respective thermodynamic properties of water were analyzed and the study was performed at the vapor–NCG interface for various operating pressures. The results indicated that the starting point of the VCHP function is significantly based on the amount of NCG content in the heat pipe and on the superheat required for boiling.

### 3.3. Thermal diodes

A simple thermal diode can be a thermosyphon in which the gravitational force supplies the irregularity when positioned appropriately. A variety of aerospace and ground based applications make use of thermal diodes which includes spacecrafts. The device is also used in modern renewable energy systems particularly where heat transfer in one direction is a requirement. However, due to the high initial capital expenditure and complexity in retrofitting such systems, commercialization and interest has increased only steadily. Fig. 7 displays the schematic of a liquid trap diode in the reverse mode [2].

Fang and Xia [11] studied the thermal performance of a novel Bidirectional Partition Fluid Thermal Diode (BPFTD) for the function of providing solar heating and passive cooling respectively. The experimental analysis was carried out by testing the BPFTD with two identical hot boxes with similar wall configuration and comparisons were established with a water-wall of optimum thickness. Test results yielded that the BPFTD had a higher heating performance compared to its water-wall counterpart with additional findings confirming an increase in heat supply of around 140% when a single glazing cover without night ventilation is utilized when compared to the water-wall respectively. Varga et al. [12] carried out tests to evaluate the performance of thermal diode panels incorporating heat pipes for passive cooling in buildings in Portugal. The manufactured experimental set-up included nine copper/water bent heat pipes with a diameter of 12.7 mm welded to aluminum sheets along with the thermal diode panels respectively. The thermal and physical properties were tested using a finite element heat transfer model combined with an optimization procedure for both forward and backward heat transfer. The work concluded the agreement of the applied model with the experimental procedure. Further, the results revealed a significant increase in the forward heat transfer results in comparison to its backward counterpart.

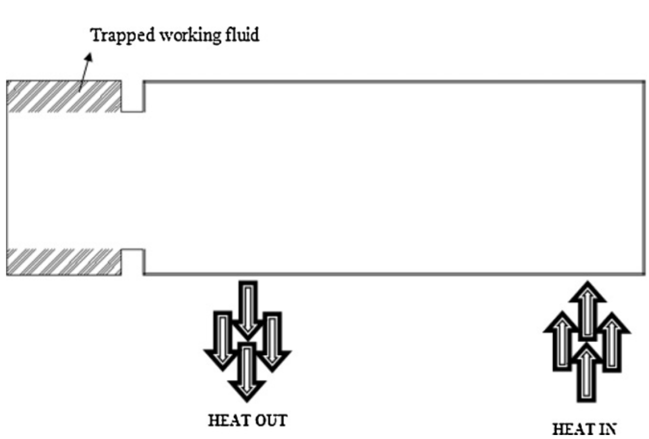



**Fig. 7.** Schematic of a reverse mode liquid trap diode.

Rhee et al. [13] experimentally investigated the temperature stratification in a solar hot water storage tank. The experiment proposed four different storage tank designs involving thermal diodes for its operation. The results of the test examined that the so-called express-elevator design displayed the highest amount of stratification during both heating and cooling periods in comparison to the other proposed designs. Consequently, the work concluded the bright future scope of optimizing the geometric parameters of thermal diodes to obtain an improved rate of stratification. Omer et al. [14] analyzed a thermoelectric refrigeration system integrated with thermal diodes to study the performance of PCMs. The fabricated system built for test included a 150 W thermoelectric refrigeration system. The performance of the proposed system was compared to another similar system without integrated thermal diodes. The results revealed the feasibility of utilizing thermal diodes between the thermoelectric cells and the PCM in order to prevent heat leakage. The results also displayed an improved performance of the system incorporating thermal diodes in the storage ability of the thermoelectric refrigeration system in comparison to its counterpart.

### 3.4. Pulsating heat pipes

A pulsating (oscillating) heat pipe consists of circuitous channel, evacuated and filled with the working fluid. Heat is transported through the latent heat of vapor and through the sensible heat transferred by the liquid slugs. When the tube on the evaporator section of the heat pipe is put under thermal load, the working fluid evaporates thus increasing the vapor pressure and formation of bubbles and transferring the liquid towards the condenser section where cooling results in a reduction of vapor pressure and condensation of bubbles in the section respectively. The increase and decrease of bubbles in the two sections lead to an oscillating or pulsating motion within the capillary tube. Qu and Ma [15] investigated the principal factors involved in startup of oscillating motions in a pulsating heat pipe including superheat and heat flux level on the evaporator section and the cavity size on capillary inner surface. The experimental investigation comprised of a glass prototype with a total length of 300 mm and the evaporator section of 90 mm along the constant inlet temperature of 296 K. The results of the theoretical analysis confirmed that the performance at startup can be improved by controlling the vapor bubble type and utilizing a rougher surface. The results also showed that the globe-type vapor bubble needs smaller superheat compared to the taylor-type vapor bubble respectively.

Wang et al. [16] studied the thermal performance of heat transport of the four-turn pulsating heat pipe by comparing various working fluids with pure water. The experimental analyses were based on two operating orientations (vertical and horizontal) of a copper tube with an external diameter of 2.5 mm. FS-39E

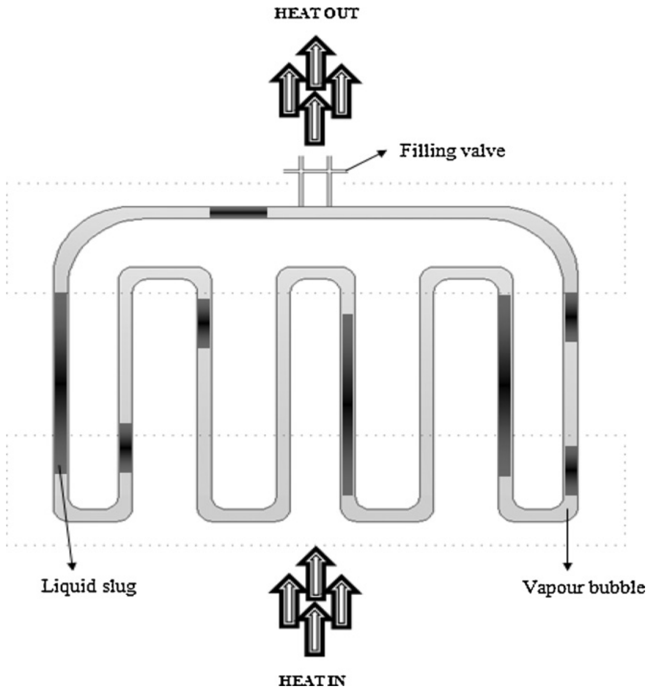



**Fig. 8.** Schematic of a pulsating heat pipe.

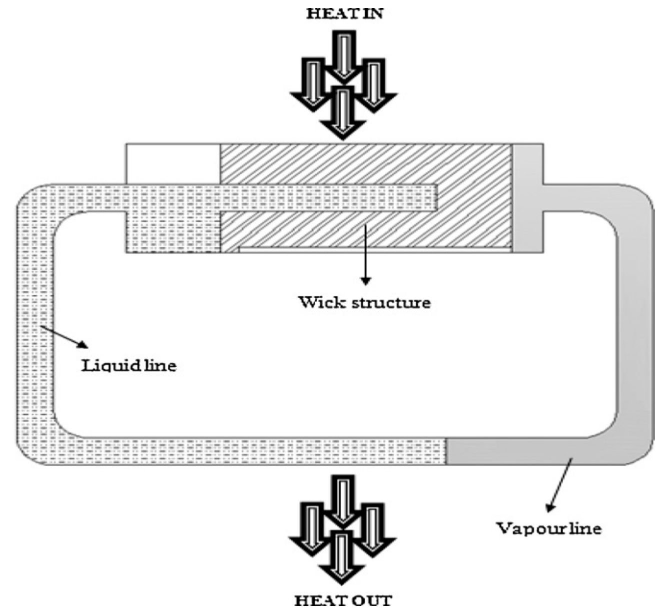



**Fig. 9.** Schematic of a loop heat pipe arrangement.

microcapsule and $Al_2O_3$ nano-fluid were used for the test. The results of the investigation proved that the functional working fluids increase the heat-transport ability of the heat pipe when compared with pure water with the FS-39E microcapsule being the best working fluid in the horizontal orientation. Fig. 8 describes the basic operation of a pulsating heat pipe [2].

Yang et al. [17] carried out work on estimating the thermal performance of closed loop pulsating heat pipes by conducting experiments on copper tubes of varying inner diameters and filling ratios respectively. The system comprised of 40 copper tubes with the inner diameters of 1 mm and 2 mm and the vertical bottom heated, vertical top heated and the horizontal orientations were compared. The investigation findings revealed that the closed loop pulsating heat pipe with the vertical bottom heating gives the best performance with 2 mm inner diameter and 50% fill ratio respectively while the orientation effects were negligible for the 1 mm inner diameter tube.

### *3.5. Loop heat pipes (LHPs) and capillary pumped loops (CPLs)*

Loop heat pipes (LHP) employ the characteristics of a conventional heat pipe but have an advantage in terms of its ability to transfer thermal energy over a larger space without any constraint on the path of the liquid or vapor lines and also in terms of a greater heat flux potential and robust operation [2]. For this reason, LHPs are fast becoming typical devices to meet the global demand of control of thermal difficulties of high-end electronics. A capillary force in the evaporator section drives the operation for the LHP requiring no auxiliary power input. Fig. 9 displays the operating principle of a loop heat pipe [2].

Wang et al. [18] conducted experiments based on a flat LHP under low-heat power input to understand the control of compensation chamber and the evaporator on the start-up behavior. The respective testing system comprised of locating the standard K-type thermocouples, DC stabilized power supply along with an isothermal cooling water tank for experimentation. The results indicated that the LHP has the potential of start-up under low heat power of 6 W. The results also confirmed that the LHP has a better start-up performance under low-power with an increasing thickness of the capillary interlayer.

Zhao et al. [19] carried out work on developing a novel LHP solar water heating system for a characteristic dwelling in Beijing in order to facilitate efficient transportation and conversion of solar heat into hot water. A numerical model was developed to monitor the overall thermal performance of the system and various parameters such as the heat pipe loop and the façade integrated solar absorber were considered for influencing results. The findings indicate that the system efficiency decreases with increasing the mean temperature of water flow and efficiency of the thermal system increases with increase in the ambient temperature. The results further confirmed that the optimum operating temperature for the heat pipe is around 345 K.

Kaya and Goldak [20] investigated the heat and mass transport in order to study the capillary porous structure of the LHP. A finite element method for the evaporator cross-section based numerical code was developed to solve the mass and energy equations and the solutions included an all-liquid and vapor–liquid wick cases. The results highlighted that at high heat loads, the boiling initiation under the evaporating meniscus is very unlikely since the liquid contact with the fin decreases significantly. The investigation concluded that in order to increase the heat transfer limit for boiling, the elimination of non-condensable gases along with a very good contact at the fin–wick interface is essential.

### *3.6. Micro heat pipes*

Micro heat pipes (MHPs) are used in applications where small to medium heat transfer rates are desirable. The rate of cooling achieved from the MHP is significantly lower compared to forced convection systems. However, the capability to control temperatures in environments of varying heat loads along with its compact size allows it to be utilized in various applications [2]. Do et al. [21] predicted the thermal performance of a flat micro heat pipe comprising of a rectangular grooved wick structure. A mathematical model was developed taking the influence of the contact angle, liquid–vapor interfacial shear stress and the amount of liquid charge. One-dimensional conduction equation for the wall and the augmented Young–Laplace equation were solved. The examined results revealed that the heat transport rate increases diffidently

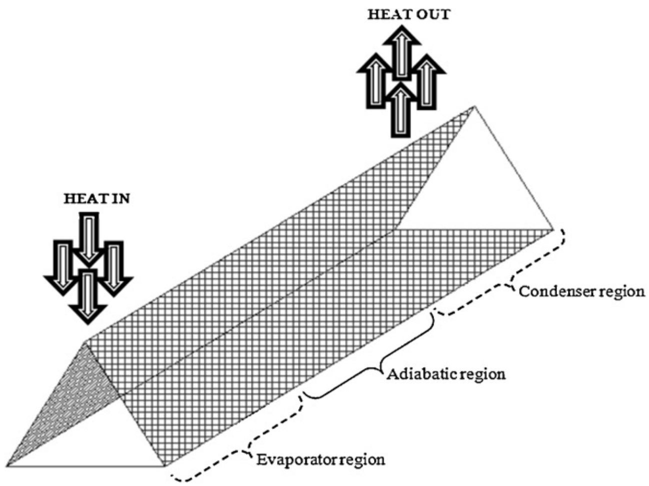



**Fig. 10.** Schematic of a micro heat pipe.

as the liquid charge increases. The findings displayed the optimization of the grooved wick structure highlighting the maximum heat transport rate of 128 W under the optimum conditions of the height and groove width respectively. Fig. 10 displays the schematic of a micro heat pipe.

Hung and Seng [22] carried out work on studying the thermal performance in terms of the heat transport capability of star-groove micro-heat pipes particularity with the influence of the geometrical design. A one-dimensional steady state numerical model was developed to solve the continuity, momentum and energy equations of the liquid and gas phases. The comparison results of the study yielded that the star-groove micro-heat pipe have a better performance characteristic compared to the conventional polygonal micro-heat pipe due to its ability to provide a higher capillary rate by the flexibility in reducing the corner apex angle. Lefèvre and Lallemand [23] investigated the heat transport capability of a flat MHP with the location of heat sources and heat sinks. A hydrodynamic 2D model containing a porous wick as a medium to behave as a capillary structure was incorporated with a 3D thermal model to study the heat conduction of both the liquid and vapor phases. The thermal model evaluated the capability to calculate the heat flux generated solely by the wall heat conductance.

### *3.7. Sorption heat pipes*

The sorption heat pipe (SHP) is a device which utilizes the sorption phenomenon on the heat pipe to improve the heat transport ability. Similar to the LHP, SHP can also be utilized in space applications since it comprises of similar evaporator and condenser along with the working fluid. Furthermore, the literature highlights that the integrity of the sorption cooler with a LHP provides higher heat fluxes and evaporator thermal resistances respectively. Fig. 11 displays the sorption heat pipe highlighting the basic components [2].

Vasiliev and Vasiliev Jr. [24] conducted an in-depth study on sorption heat pipes as a heat transfer device and highlighted the potential in order to be utilized in cryogenic fluid storage due to its high heat transport ability. The investigation was based on an experimental set-up, comprising of a sorption cooler and a capillary pumped evaporator for both sorption and loop heat pipe arrangement. The results of the experiment revealed that the heat transfer by the sorption heat pipe was in excess of 12 kW/(m$^2$ K), an increase of three times in comparison to a loop heat pipe respectively.

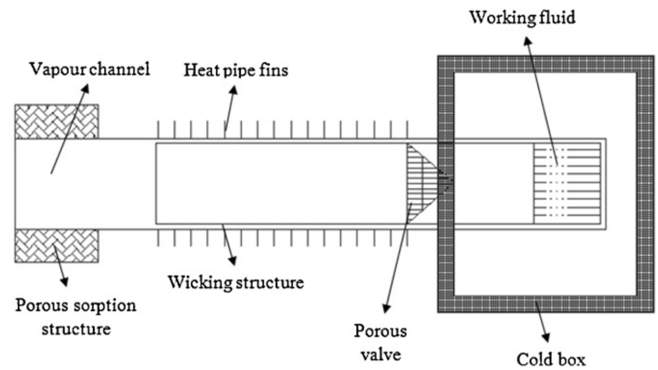



**Fig. 11.** Schematic of a sorption heat pipe.

## 4. Computational simulation associated with heat pipes

From extensive literature, computational studies developed on various heat pipe arrangements, displaying two-phase flow patterns highlight the broad scope of the respective technology for use in various passive and active applications as reviewed earlier. Viable numerical codes have developed into useful tool for determining specific and precise results for the overall performance of various multiphase flow patterns and phase change behaviors respectively.

Alizadehdakhel et al. [25] studied the operation of a thermosyphon by simulating two-phase flows using FLUENT 6.2 commercial CFD code and validating the results with an experimental set-up using various operating parameters. A two-dimensional geometry was modeled using the Gambit software with the domain consisting of a total number of 47,124 and 14,361 grids for the fluid and the solid region respectively. The Volume of Fraction (VOF) [26] method was established for the two-phase flow modeling. Various heat flux values obtained from the experiments were applied as the energy inlet to the evaporator and a vapor pressure of 1.72 kPa at 288 K was applied to water in the gaseous phase. A good agreement was achieved between the CFD and experimental temperature profiles across the length of the pipe. The experimental results confirmed that increasing of the inlet heat flow from 350 to 500 greatly enhances the thermosyphon's overall performance. The conclusions of this study confirmed that the complex heat and mass transfer phase changes can be effectively modeled and a greater perceptive of the phase change is observed using CFD.

However, a range of numerical codes have been applied in order to develop a precise understanding of the two-phase behavior inside a heat pipe. Lin et al. [27] investigated the potential of utilizing heat pipe heat exchangers for use in dehumidification processes to understand the performance of the system. The FLOTHERM numerical code in conjunction with Microsoft Excel commercial package was used for CFD simulation of a drying cycle in the dehumidification process using characteristic air properties with an inlet temperature variation between 308 and 323 K along with a relative humidity of 100% and a volume flow rate variation between 6 and 8 L/s. The heating and condensing regions were defined in the domain for the calculation of fluid parameters and properties for the simulation. Values for the cuboids representation of heat pipes with thermal conductivity, specific heat capacity and density were obtained. The predicted results confirm the performance of the system at various operating conditions and show that a significant improvement in dehumidification process is possible using the heat pipe solution with higher condensate rates obtained at higher inlet flow rates and temperatures. However, the results further confirm that the heat transport in the heat pipe decreases with increasing flow rate signifying the potential of a heat exchanger utilizing auxiliary power can work with better efficiency at higher flow rates.

Ranjan et al. [28] conducted numerical analysis on the study of flat heat pipes or vapor chamber by solving the vapor and liquid flow using three-dimensional Navier–Stokes continuity, momentum energy equations to understand the effect of varying wick microstructure on evaporation and condensation sections of the heat pipe. Temperature and flow contours were computed by a device-level numerical macro-model unaided and coupled with wick-level micro-model to account for the evaporation heat transfer rate in the pores of general sintered-powder wick structures using the commercial FLUENT solver. The coupled model incorporates corrections to the evaporative mass flow rates at the liquid–vapor interface based on local contact angle of liquid in the wick. An effective conductivity value of 40 W/(m K) was assumed for the macro-model while the convective heat transfer boundary conditions for the micro-model consisted of a constant inlet temperature and pressure to investigate the liquid meniscus between copper wires. The results based on the two models (coupled and non-coupled) revealed that the thermal resistance by the liquid–vapor interface increases affecting the performance of the vapor chamber as the device is decreased in magnitude.

Shao and Riffat [29] investigated the performance of a heat recovery system based on a heat pipe arrangement at different positions inside passive stacks for natural ventilation systems. The FLUENT solver was used for CFD coding to simulate flow losses in the ventilation stack by solving the mass and energy conservation equations respectively. The domain mapped by a uniform Cartesian grid of $50 \times 100$ comprised of the two-dimensional geometry of the exhaust stack and the building space to understand the buoyancy flow in the room. The boundary conditions involved a constant external and internal stack wall temperature of 288 and 293 K. The findings from the computational simulation displayed that the average vertical velocity in the stack is 0.223 m/s along with a pressure differential of in excess of 29 Pa obtained between the inlet and outlet respectively. Further, the investigation proved that the insertion flow loss is higher when the heat pipes are located at the bottom of the vertical stack compared to the top and is inversely proportional to the insertion pressure loss. It was noted that the heat pipes did not cause a significant reduction of stack flow.

Saber and Ashtiani [30] developed a computational fluid dynamic model to optimize the distribution of fluid flow parameters in order to find its influence on the evaporator performance inside the heat pipe heat exchanger. Compressible flow equations were solved for the numerical simulation using the FLUENT commercial code. The geometry comprised of 6 rows of 12 tubes each and the temperature was set to 793 K. Further boundary conditions included an inlet mass flow rate of 3.75 kg/s. Four distinct cases were analyzed for distributed flow with Case 1 being nominal conditions. Case 2 included the cross section area of the inlet area to be doubled. Case 3 uses an additional horizontal plate after entry while Case 4 uses an imperfect cone at the entry. The cases were used to highlight the output temperature differential profiles in order to highlight the maximum efficiency potential. The results of the extensive study revealed that better flow distributions are possible with an increase in cross section area of the inlet but also increases operation costs and pressure drop respectively. The results further highlight the use of baffles coupled with the imperfect cone has positive impact on the flow distribution and produces optimum efficiency.

Rahmat and Hubert [31] developed a triangular two-phase model of a micro-heat pipe to study the heat and mass transfer inside the three-dimensional micro channel. Ansys CFX-5.7.1 commercial software was used for solving the unsteady flow equations. The channel geometry was divided into three identical portions to incorporate the evaporator and condenser section behavior. The length of the evaporator and condenser section was 0.67 cm respectively. The meshed model comprised of 560,000 elements while the average working fluid volume of the elements was 310 $\mu m^2$. The fluctuation of convergence results with respect to various fill ratios and boundary condition type was investigated for precise performance. The findings showed that the effective thermal conductivity of 3333 W/°C was obtained for the micro channel at a fill ratio of 25%. Further, the results concluded that an increase in liquid fill ratio causes an increase in the effective length of the heat pipe. The investigation confirmed a good agreement between the computational findings with relevant literature, highlighting the capability of commercial finite element codes in order to successfully simulate two-phase flows.

Thermal effectiveness of experimental procedures incorporating heat pipes has increased over the years with the introduction of thermal imaging systems. Hemadri et al. [32] conducted an extensive study on the feasible utility of pulsating heat pipes in thermal radiator systems for terrestrial and space applications. An understanding of temperature profiles were developed experimentally by using a high-resolution, forward looking infra-red camera for varying thermal and mechanical boundary conditions. The experiment was conducted on aluminum and mild steel radiator plates with and without embedded pulsating heat pipe arrangement aligned in three distinct orientations. Surface mounted flat mica heater of known dimensions was used for heat generation at varying thermal input between 50 and 150 W. The outcome drawn from the investigation included spatial thermography and the effects of orientation respectively. It was observed that the pulsating heat pipe arrangement provided limited improvement to the rate of isothermalization due to the high base thermal conductivity of the aluminum plate. The results further displayed the increase in domination of gravitational forces at low heat input of for the vertical orientation with heater position upwards for both plates. It was concluded that the gravitational effects were reduced increasing the pulsations with increasing thermal input. The experimental results were validated using the FLUENT 6.3.26 commercial code using the three dimensional tetrahedron computational domain. A good agreement was therefore observed between the experimental and simulated temperature profiles at a heat input of 55 W for various locations across the plate on a unit-cell model. The work highlighted the potential of pulsating heat pipes in efficient thermal management for space and terrestrial sectors.

Savino et al. [33] investigated the effect of surface tension variation with temperature to highlight the performance of self-rewetting fluids in comparison to ordinary fluids in wickless heat pipe systems. Temperature profiles using thermographic images were developed by conducting laboratory experiments on glass tubes containing alcohol and 1-heptanol aqueous solution respectively. The apparatus included an infra-red thermal imaging camera and thermal power was kept between 4 and 7 W to limit the evaporation phenomenon. The bubble trajectory displayed that the linear movement is in the direction of the temperature gradient for an ordinary fluid and vice versa for the self-rewetting fluid. Navier–Stokes equations were solved using the SIMPLE family of algorithms and Volume of Fraction (VOF) model in the FLUENT commercial code was used for computational investigation in order to validate the experimental results. Further similar laboratory tests were performed to establish the surface tension gradient in relation to the temperature variation. It was observed that the ordinary fluid (ethanol) exhibited a decreasing linear dependency on the temperature while the self-rewetting fluid (heptanol) showing a non-linear dependence. The detailed study emphasized the potential of efficient heat transfer by introducing new working self-rewetting fluids on the binary mixtures based on Water/Ammonia and Water/Ethylene Glycol for various applications.

**Table 1**
Summary of the heat pipe technologies under review.

| Type | Features | Limitations | Applications | Industrial equipment | Operating temp. (K) | Refs. |
|---|---|---|---|---|---|---|
| Tubular heat pipe | Simple and effective passive operation. | Requires clean air stream for optimum operation. | Injection moulds and air to air heat pipe heat exchangers. | Thermacore Copper–Water | 218–453 | [2,34] |
| Variable conductance heat pipe (active control) | Superior heat source and temperature control. | Supplementary power requirement. | Accurate satellite temperature calibration and removal of heat from radioactive waste. | Thermacore VCHP with heated reservoir | 268–338 | [2,35] |
| Thermal diode | Unidirectional heat flow. | Complexity in retrofitting the system. | Gamma-ray spectroscopy and collection of solar gain for space heating. | National semiconductor Dual Thermal Diode Sensor | 273–398 | [2,35] |
| Pulsating heat pipe | Growth and collapse of working fluid provides the driving force. | Increased cost and weight due to inflexible metallic pipe material. | Electronic and central processing units cooling systems. | Sun Microsystems PHP | 273–378 | [2,36] |
| Loop heat pipe | Entrainment is minimized by separate wick and liquid flow paths. | Supplementary power requirement for mechanically pumped loops. | Solar water heating and control systems for military aircrafts. | Thermacore Ammonia LHP | 200–400 | [19,37] |
| Micro heat pipe | Iso-thermalization and flaccid operation. | Inferior heat transfer ability compared to its counterparts. | Cooling laser diodes and thermal control of ceramic chip carriers. | Furukawa High-Performance μHP | 313–343 | [2,38] |
| Sorption heat pipe | Convective heat transfer by integrity with a sorption machine in a single unit. | Intricate cycle compared to a simple heat pipe. | Cryogenic fluid storage. | Luikov Heat and Mass Transfer Institute SHP | 60–400 | [24] |

## 5. Results summary

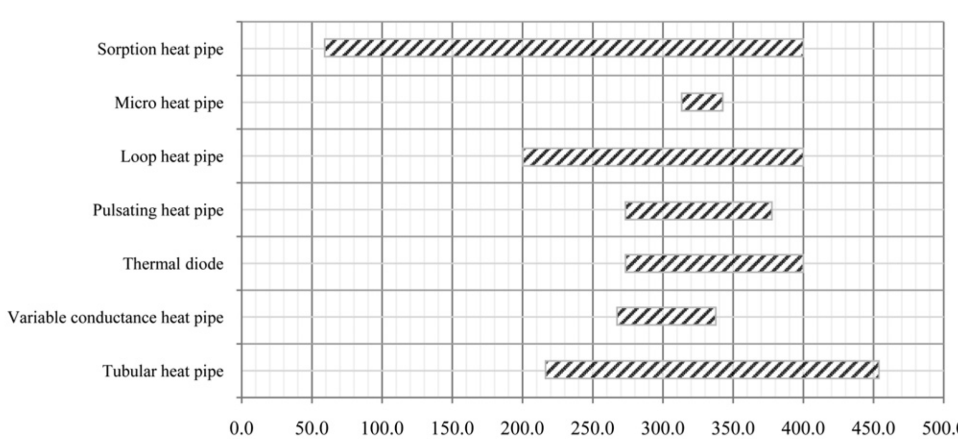
From the reviewed heat pipe technologies for terrestrial and aerospace applications, it is considered that each system has its own advantages and limitations based largely on working conditions. Table 1 summarizes the reviewed heat pipe systems displaying the typical applications and range of operating temperatures. The figure of merit is estimated based on the working fluid properties based on operating temperatures formulated in Eq. (1).

$$M = \frac{\rho_l \sigma_l L}{\mu_l} \qquad (1)$$

where $\rho_l$ is the density of liquid; $\sigma_l$ is the surface energy per unit area of liquid; $L$ is the latent heat of vaporization; $\mu_l$ is the dynamic viscosity of liquid.

## 6. Discussion

With reference to Table 1, the principal properties and applications were obtained based on relevant commercial equipment. As observed, the working range for tubular heat pipe systems is at intermediate temperatures with average operating temperature being 335.5 K. The industrial manufacturers for the respective highlight the ability of the copper–water sintered-powdered wicked heat pipe device to transfer thermal energy efficiently regardless




| | Tubular heat pipe | Variable conductance heat pipe | Thermal diode | Pulsating heat pipe | Loop heat pipe | Micro heat pipe | Sorption heat pipe |
|---|---|---|---|---|---|---|---|
| Average Operating Temp. (K) | 335.5 | 303.0 | 335.5 | 325.5 | 300.0 | 273.0 | 230.0 |

Average Operating Temp. (K)

**Fig. 12.** Operating temperature range comparison of the reviewed heat pipe systems.

**Table 2**
Merit No. for various working fluid candidates at operating temperatures.

| Medium | Melting point (K) | Boiling point (K) | Useful range (K) | Merit No. at operating temp. (293 K) | Merit No. at operating temp. (313 K) | Merit No. at operating temp. (333 K) | Merit No. at operating temp. (353 K) | Merit No. at operating temp. (373 K) | Merit No. at operating temp. (393 K) |
|---|---|---|---|---|---|---|---|---|---|
| Heptane | 183.15 | 371.15 | 273–423 | 1.16E+10 | 1.25E+10 | 1.24E+10 | 1.24E+10 | 1.19E+10 | 1.10E+10 |
| Water | 273.15 | 373.15 | 303–473 | 1.78E+11 | 2.55E+11 | 3.27E+11 | 3.90E+11 | 4.55E+11 | 4.97E+11 |
| Ammonia | 195.15 | 240.15 | 213–373 | 7.02E+10 | 5.85E+10 | 4.50E+10 | 2.30E+10 | 1.45E+10 | 3.43E+09 |
| Pentane | 140.15 | 301.15 | 253–393 | 1.49E+10 | 1.48E+10 | 1.35E+10 | 1.22E+10 | 1.03E+10 | 7.76E+09 |
| Acetone | 178.15 | 330.15 | 273–393 | 3.20E+10 | 3.24E+10 | 3.17E+10 | 3.00E+10 | 2.57E+10 | 1.31E+10 |

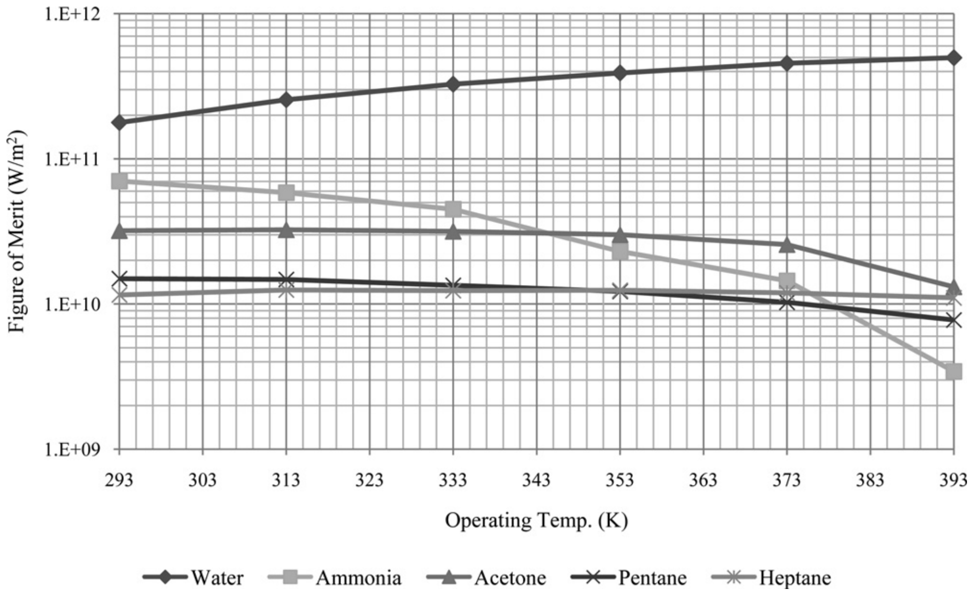



**Fig. 13.** Merit No. of candidate heat pipe working fluids for intermediate temperatures.

of orientation and gravitational effects with the density estimation of 50 W/cm$^2$. Other imperative features of tubular heat pipe systems include compactness, integrity into heat sinks and cold plates through mechanical interference and long-life reliability which is a highly desirable factor for a majority of terrestrial applications.

A graphical representation of the operating temperature range of reviewed heat pipe systems is displayed in Fig. 12. It is seen that the range of working temperatures is maximum for SHP systems highlighting its superiority to replace SHP systems in space applications. Moreover, from extensive literature, it is asserted that the integration of the sorption cooler with LHP systems has recorded average evaporator thermal resistances of 0.07–0.08 K/W with heat fluxes of 100–200 W/cm$^2$ [2]. Further observations from Fig. 12 include the temperature range limitations for micro heat pipe systems which has a differential of only 30 K since nearly all of the high-performance utility includes enhancing heat transfer of electronic components, namely computer central processing units and microprocessors which operate at working temperatures of 313–343 K.

Typical operating temperatures for heat pipe systems utilized in terrestrial applications range from 293 K to 393 K. Choice of working fluid is a major contemplation in identifying appropriate heat pipe assemblies and candidate working fluids are summarized in Table 2 for intermediate temperatures. The Merit No. is a useful indicator in determining the maximum heat transport capability in terms of the fluid properties and is determined by Eq. (1).

Fig. 13 displays the Merit No. variation with increasing intermediate operating temperatures for a range of heat pipe working fluids. With reference to the figure, a significant increase in Merit No. of 64.2% for Water is observed while notable decreasing gradient of 95.1% and 59.0% is observed for Ammonia and Acetone with increasing temperatures respectively. As expected, water demonstrates a much superior Merit No. in comparison to other candidate fluids within the operating temperature range, thus confirming the historical dominance as the principal working fluid in most heat pipe applications.

## 7. Conclusion

The technological development of research into the utilization of heat pipes for efficient and passive heat transport is rapidly increasing through the use of advanced computation and complex experimentation techniques. This study reviewed some of the general heat pipe systems used in building and ground applications including heat recovery and renewable energy methodologies in order to determine the typical heat pipe arrangements along with their working temperature range for use in the respective. The investigation revealed that heat pipes incorporated with sorption phenomenon display greater heat transfer capacity and tubular heat pipes have the highest working range on average with the maximum operating temperature from all reviewed systems being 453 K for the tubular heat pipe arrangement respectively.

The study's conclusions are based on the research of various industrial products utilizing the heat pipe systems for their operations. Imperative factors including the figure of merit were calculated and compared for various suitable heat pipe working fluids. The findings revealed that water displayed the highest

average Merit Number in comparison to ammonia and acetone for the operating temperature range of 293–393 K.

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